A detailed FEM study on the vibro-acoustic behaviour of crash and splash musical cymbals

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Abstract: Advanced numerical simulations, that include modal and frequency response function finite element analysis, frequency domain and time domain finite element method - boundary element method analysis, are performed to study the vibro-acoustic behaviour of crash and splash musical cymbals. The results of the modal analysis agree well with experimental measurements found in literature. The frequency domain and time domain coupled finite boundary element method simulations, despite their high computational resources and time demands, are used for the crucial comparison of the velocity spectrograms on the cymbal to the radiated sound pressure spectrograms in the air. The computational analysis results show that the splash cymbal is characterized by a faster decay and a higher frequency content compared to the crash cymbal. The advanced multiphysics vibroacoustic simulations that correlate the displacements and velocities of the vibrated structure with the radiated sound pressure results demonstrate the future capability to synthesize the sounds of cymbal music instruments.

Key-Words: vibro-acoustics, finite element method, boundary element method, musical instruments, simulations

I. INTRODUCTION

Cymbals are ancient idiophone musical instruments which are stimulated by percussion. They are played by hitting them with a drumstick or with another cymbal. Their basic form is that of a metal hollow circular disk slightly elevated in its center. Cymbals are mainly made of bronze or other alloys in various diameters and thicknesses [1-3]. The sound produced by cymbals is inharmonic and thus does not have a perceivable pitch. It is of broad spectrum which is remarkably rich in high frequency content. There are different types of cymbals used by music ensembles of differed size (i.e., orchestral, bands or small groups) and different music genre (i.e., classical, jazz, traditional). Examples of cymbal types are the hi-hat, the ride, the crash, the splash, and the China cymbals. Cymbals usually range from 20 to 75 cm in diameter. When discussing the anatomy of a cymbal, there are three physical areas that, when struck, produce different vibration patterns and corresponding frequency responses, which determine the overall sound of the cymbal. These are the bell (the raised area at the center of a cymbal), the edge (the order periphery), and the bow or body (the curved area between the bell and edge). The bell seems to play a central role in determining the overall sound of a cymbal. Specifically, while cymbals that are thinner at their edges are more 'crushable' and have a faster response than those with thicker edges. Overall, the sound of a cymbal is affected by the size of the bell, the shape of the bow and its diameter, the total weight, and the material of the cymbal [1,4].

The works found in literature focusing on the vibrating behaviour of idiophone cymbals are rather limited. Using holographic interferometry Rossing et al [5] measured the vibration modes of a 38 cm diameter cymbal and observed that the first six modes resemble those of a flat plate, while at higher frequencies, the modes of vibration often mix with one another, and hence mode identification is not straightforward. Wilbur et al [6] measured more than 100 modes of vibration in a 46 cm diameter crash cymbal using electronic TV holography. Perrin et al. [7,8] identified more than hundred normal modes of cymbals using various experimental methods such as electronic speckle pattern interferometry, laser vibrometry, and patterns and compared the Chladni sand experimental results with finite element (FE) numerical simulations. The results showed that certain modes couple with other modes when they are close in frequency. Kuratani et al [4] studied the effect of the hammering process on the vibration characteristics of cymbals via FE simulations. Nguyen et al [9] studied the vibrations of thin plates and shells with variable thickness via analytical modelling and FE simulations with the purpose to synthesize the sounds of cymbal-like instruments. Moreover, it was shown that at high vibration amplitudes, the strong aftersound that provides cymbal sound its characteristic shimmer involves highly nonlinear processes [2,10]. These processes seem to occur when the amplitude of the vibration is higher than the thickness of the cymbal [11]. Recently, Samejima [12] extended the physical modeling of cymbals including the dynamics of washers supporting the center of cymbals and sticks/mallets striking the cymbals using coupled finite difference and FE simulations.

The aim of the present study is to study the vibrational behaviour of a 48.3 cm (19 in.) crash and 25.4 (10 in.) splash cymbals via detailed simulations that include modal and frequency response function (FRF) FE analysis, frequency domain Finite Element Method - Boundary Element Method (FEM-BEM) analysis and time domain transient FEM-BEM analysis. FEM is selected since is it versatile due to its flexibility in modeling complicated geometries when the domain changes, when the desired precision varies over the entire domain or when the solution lacks smoothness [13-16]. The simulation results are compared to each other as well as to experimental data found in the literature. It is of great importance that, to the authors knowledge, this is the first study comparing the simulation results of velocity spectrograms on the cymbal with the radiated sound pressure spectrograms, in the air, via the frequency and time domain transient FEM-BEM simulations. These simulations demand high computational resources and runtime.

II. MATHEMATICAL DESCRIPTION

FEM is ideal for predicting how musical instruments react to any kind of force loads, vibrations, and variations in environmental conditions (temperature, relative humidity, etc.) [17,18]. Here, modal and FRF FEM simulations, frequency domain FEM-BEM simulations are performed. The basic equation of motion (for structural analysis) which is numerically solved is:

$$[M]\left\{\frac{\partial^2 U}{\partial t^2}\right\} + [C]\{\dot{U}\} + [K]\{U\} = \{F\}$$
(1)

where [*M*] is the mass matrix, [*C*] is the damping matrix, [*K*] is the stiffness matrix, {*U*} is the displacement vector and {*F*} is the load vector. The damping in the system may be defined through a stiffness matrix multiplier used to form the viscous damping matrix as $[C]=\beta[K]$, where β is the Rayleigh damping constant.

A. Modal and FRF analysis

Ignoring external forces, for harmonic motion in the frequency domain, Eq.1 results in a modal eigenvalue problem of the following form:

$$([K](1 + \omega\beta \operatorname{cot}(\omega t)) - \omega^2[M])[\Phi] = 0 \qquad (2)$$

where $[\Phi]$ is the modal matrix, whose columns are eigenmodes and ω are angular eigenfrequencies.

FRF is computed using the mode superposition method, in the frequency domain. Considering a $\{p(t)\}$ external force, Eq.1 takes the form:

$$[M]\left\{\frac{\partial^2 U}{\partial t^2}\right\} + [C]\{\dot{U}\} + [K]\{U\} = \{p(t)\}$$
(3)

Using the mode superposition method, the displacements response is expressed as:

$$\{U\} = \sum_{n=1}^{N} \varphi_n q_n(t) \tag{4}$$

where φ_n and $q_n(t)$ are the n^{th} mode shape and modal coordinates, respectively. For a *N*-degrees of freedom (DOF) system and for each of the *N* differential equations in modal coordinates in the frequency domain, it holds:

$$q_n(\omega) = \frac{p_n(\omega)}{(-\omega^2 + i\beta\omega_n^2\omega + \omega_n^2)M_n}$$
(5)

Substituting Eq.5 into Eq.4, the structural displacement response in the frequency domain becomes:

$$\{U(\omega)\} = \sum_{n=1}^{N} \frac{\varphi_n}{(-\omega^2 + i\beta\omega_n^2\omega + \omega_n^2)M_n} p_n(\omega) \qquad (6)$$

Supposing that the excitation was applied at node j and the response is evaluated for node k, the acceleration frequency response function F_a can be expressed as:

$$F_{a} = -\omega^{2} \sum_{n=1}^{N} \frac{\varphi_{n}(x_{k})}{(-\omega^{2} + i\beta\omega_{n}^{2}\omega + \omega_{n}^{2})M_{n}} P_{n}\left(x_{j}\right)$$
(7)

where $P_n(x_j) = \varphi_n^T p(x_j)$, and $p(x_j)$ is the space distribution of the harmonic force excitation.

B. Frequency domain FEM-BEM and time domain FEM-BEM analysis

The frequency domain response is computed utilizing a harmonic response analysis, which is employed to determine the steady-state dynamics of the vibrated structure in response to loads that vary harmonically with time. The excitation spectrum can be given as nodal force, as pressure or as base acceleration, and considers complex variable input. FEM steady-state dynamic (SSD) analysis results may be computed using the mode superposition method like in the case of the FRF analysis. This case differs since the external force in Eq.3 varies harmonically with time. Moreover, the transient dynamic analysis may also be used to determine the response of the structure, which is subjected to a time-dependent loading, considering inertia and damping effects. In that case, time dependent force or pressure or displacement, may be considered as the excitation source of the dynamic problem in Eq.1.

In this work, LS-DYNA [19] solvers perform the modal and FRF, the frequency domain and the time domain FEM-BEM analysis. LS-DYNA has the capability to provide an integrated solution for vibro-acoustic problems by coupling the FEM transient dynamic solver with a BEM acoustic solver. The BEM method is used to model the environment that surrounds the vibrated structure and calculates the pressure of the radiated sound [20]. Regarding the frequency domain FEM-BEM analysis, an SSD analysis is carried out, based on the results of a modal analysis (frequencies, modal

shapes), to provide the vibrating response of the structure in the frequency domain. The obtained boundary velocities from the SSD analysis provide boundary velocities for the subsequent BEM acoustic computations. For the time domain FEM-BEM analysis, initially the time domain FEM analysis is performed, and the time domain dynamic response of the structure is converted to frequency domain by using the Fast Fourier Transform (FFT). Then, likewise with the frequency domain FEM-BEM analysis, the obtained boundary velocities provide boundary velocities for the subsequent BEM acoustic computations [21].

For the BEM frequency domain acoustic analysis, the acoustic wave propagation in an ideal fluid, with no presence of any volume source, is governed by the Helmholtz equation:

$$\nabla^2 p + k^2 p = 0 \tag{8}$$

where p is the acoustic pressure and k is the wavenumber. By using Green's theorem Eq. 8 can be transformed into an integral equation. In this case, the pressure at any point in the fluid medium can be expressed as an integral of surface pressure and surface velocity of a vibrating structure by the following equation:

$$P_Q(\omega) = \int_S \left(G \frac{\partial p(\omega)}{\partial n} - p(\omega) \frac{\partial G}{\partial n} \right) dS \qquad (9)$$

where $P_Q(\omega)$ is the sound pressure at field point Q, S is the structure surface, $p(\omega)$ is the surface pressure, n is the normal vector on the surface S and G is the Green's function, which is equal to:

$$G = \frac{e^{-ikr}}{4\pi r} \tag{10}$$

where, r is the distance between the field point Q and the surface integration point. It also holds that:

$$\frac{\partial p(\omega)}{\partial n} = -i\rho\omega v_n \tag{11}$$

where ρ is the acoustic fluid density and v_n is the normal velocity [22]. Therefore, the knowledge of pressure and velocity on the surface allows for the pressure determination at every field point.

III. MODELING AND SIMULATION

For all the different types of simulations the CAD geometries of a 48.3 cm (19 in.) crash and 25.4 cm (10 in.) splash Zidjian[®]-type cymbals were developed and are shown in Fig.1. The diameter of the bell for the crash cymbal is 95 mm and its inner diameter is 10 mm, while the diameter of the bell

for the splash cymbal is 83 mm and its inner diameter 15 mm. The thickness from the centre to the edge of the cymbal varies and the thickness at the edge of both cymbals is 1 mm.



Fig.1 CAD geometries of a) the crash cymbal and b) of the splash cymbal.

The mesh of the geometry consists of approximately 28000 and 15000 shell elements, for the crash and the splash cymbals respectively, generated after a mesh independent study, while a quadrilateral shell element is used for the simulations.

The material properties of bronze are set according to [4]. The density is 8700 kg/m³, the Young's modulus is 85.1 GPa and the Poisson's ratio is 0.36. The Rayleigh damping constant is 0.001 [4]. Regarding the boundary conditions, a clamped constraint is imposed on the nodes bounding the central hole, simulating the cymbal's attachment to a supporting structure [7,23].

For the SSD analysis, the loading function $F(\omega)=90\cos\omega t$ (in N) is imposed as the excitation impact force. This force acts on a selected node at the edge of the cymbals. For the transient dynamic analysis, a strike is given as an input force to the cymbal. The time dependent force has the form [9]:

$$F(t) = \begin{cases} \frac{p_m}{2} \left[1 + \cos\left(\frac{\pi t}{T_{wid}}\right) \right], t \le T_{wid} \\ 0, t > T_{wid} \end{cases}$$
(12)

where T_{wid} is the temporal width of the interaction and p_m is the amplitude of the force. A short interaction time of $T_{wid}=1$ ms is selected to mimic a strong hit given by a drumstick, while the amplitude of the strike p_m is 90 N [9].

For both frequency domain and time domain FEM-BEM acoustic simulations a frequency range of 0-4000 Hz is considered. The upper limit is selected based on experiments of the measured dominant modes of vibration for a crash and a splash cymbal [7.8]. The output frequencies are also set to this range, while a resolution of 1 Hz is considered. The acoustic medium is air at room temperature with a density of 1.21 kgm⁻³, a speed of sound of 340 ms⁻¹ and a reference pressure of 20 μ Pa. The geometry of the air sphere that contains the drumhead consists of 2000 BEM elements. Moreover, a massless acoustic node is placed 0.5 m in front and 0.5 m above the center of the cymbal. This node corresponds to an assumed microphone position in the acoustic field, where the sound pressure from the impact is measured.

IV. RESULTS AND DISCUSSION

The results of the FEM modal analysis of the first 8 modes of vibration for the crash cymbal are shown in Fig.2a. The modes are designated by (m, n), where m is the number of nodal diameters and n is the number of nodal circles. These results are in a good agreement compared to experimental results of a crash 45.7 cm (18 in.) diameter cymbal found in [8]. The results of the first 8 modes of vibration for the splash cymbal are also shown in Fig.2b. Due to the smaller size of the splash cymbals the computed frequencies are higher compared to the crash cymbal. This fact agrees with the work in [8], where experimental results of a 30.5 cm (12 in.) splash cymbal present higher frequencies compared to the 18 in. crash cymbal. For these modes the response at each resonance was dominated by one mode. The deflection shape at each resonance frequency is approximately equal to the mode shape. However, at higher frequencies, the modes of vibration often mix with one another since they have almost the same natural frequency, and mode identification becomes nontrivial. The deflection shape at each peak does not coincide with any of the mode shapes; it is rather a combination of their mode shapes [3,4]. Nevertheless, a detailed investigation of the mixing of the modes is beyond the scope of this research study.



Fig.2 FEM modal analysis results for the first 8 modes of a) the crash cymbal and b) of the splash cymbal.

Fig.3a and b show the results of the FEM FRF analysis for the crash and the splash cymbals, respectively. The amplitude ratio of the output acceleration to input excitation force is shown in relation to frequency. In Fig.3, the mode (2,0) corresponds to M1, the mode (3,0) to M2, the mode (4,0) to M3, the mode (5,0) to M4, the mode (6,0) to M5, the mode (7,0) to M6, the mode (8,0) to M7 and the mode (2,1) to M8. The results of the FRF analysis agree with the modal analysis and it is apparent that the splash cymbal presents higher frequencies for the same modes of vibration. This agrees with the fact that the generated sound of the cymbal consists almost entirely of high frequency content [1,23]. In the splash cymbal mixing modes appear between the eight first modes. The mixing mode (6,0) + (3,1) appears at 819 Hz, while the mixing mode (8,0) + (4,1) appears at 1033 Hz. This mode exists also in the crash cymbal at 452 Hz.



Fig.3 FEM FRF analysis results of a) the crash cymbal and b) of the splash cymbal.

Fig.4a and b present results of the frequency domain FEM-BEM analysis for the crash and splash cymbals, respectively. The normalized acoustic pressure, in logarithmic scale, is shown in relation to the frequency, while the eight first modes of structural vibration are identified in the pressure graph. It is interesting that in Fig.3a the peaks of the M6 and M8 modes of vibration have become valleys of pressure in Fig.4a for the crash cymbal. Likewise, in Fig.3b the peaks of the M3, M5 and M7 modes of vibration have become valleys of pressure in Fig.4b for the splash cymbal.



Fig.4 Frequency domain FEM-BEM results of sound pressure in relation to frequency of a) the crash cymbal and b) of the splash cymbal.

Fig.5a, b and c show results of the time domain FEM-BEM analysis. Fig.5a and b demonstrate the radiated normalized sound pressure at the field point for the crash and the splash cymbals respectively, while Fig5.c demonstrates the normalized displacement for the splash cymbal. Comparing the results of Fig.5a and b it is observed that the oscillations of the sound pressure are attenuated faster for the splash cymbal. This comes in line with the fact that the splash cymbals have a faster decay [24]. Moreover, it is noticed that the oscillations of pressure decay faster compared to oscillations of displacement, shown in Fig.5c.



Fig.5 Time domain FEM-BEM results of a) pressure for the crash cymbal, b) pressure for the splash cymbal and c) displacement for the splash cymbal.

Furthermore, Fig.6 shows time domain FEM-BEM results, corresponding to Fig.5, that concern the pressure spectrogram for the crash cymbal in Fig.6a, the pressure spectrogram for the splash cymbal in Fig.6b and the velocity spectrogram for the splash cymbal in Fig.6c. By comparing the results of Fig6a and b, it is clear that the splash cymbal has higher frequency content, compared to the crash cymbal. The results of the pressure and velocity spectrograms in Fig6.b and c present a lot of similarities, however the harmonics developed on the vibrating structure are clearer and more discrete in relation to the harmonics developed on the air. It is for the first time, to the authors knowledge, that such a comparison is made since the performed time dependent FEM-BEM simulations are time consuming and demand very high computational resources. This was achieved due to the use of the High-Performance Computer (HPC) ARIS for parallel processing [25].



Fig.6 Time domain FEM-BEM results: a) pressure spectrogram for the crash cymbal, b) pressure spectrogram for the splash cymbal and c) velocity spectrogram for the splash cymbal.

V. CONCLUSION

Numerical simulations were performed for the study of the vibro-acoustic behaviour of crash and splash musical cymbals. In summary we conclude that:

- The modal analysis results agree well with the experimental measurements found in the relevant literature.
- The frequency domain FEM-BEM simulations results of acoustic pressure are in line with the FRF results of the vibrated structure.
- For first time velocity spectrograms on the cymbals are compared to radiated sound pressure spectrograms, in the air, via the frequency and time domain transient FEM-BEM simulations, performed on the (High

Performance Computer) HPC ARIS for parallel processing.

- The splash cymbal decays faster and exhibits significant energy at higher frequencies (above 1kHz) compared to the crash cymbal.
- The simulations results demonstrate the perspective of synthesizing different cymbal sounds parametrized on their physical attributes, which will be further investigated.

Future work will focus on experimentally validating the results of the numerical simulations via pressure measurements of selected physical cymbals, therefore permitting to safely associate physical parameters of the model with the reproduced sound field.

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